The Investigation of Heat Transfer Performance and Fluid Flow Characteristics of Titanium Oxide-Water Nanofluid in a Shell and Tube Heat Exchanger with Inclined Baffles

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ABSTRACT.

This work presents a three-dimensional analysis for a shell and tube heat exchanger with inclined baffles where Titanium oxide-water nanofluid is the cooling fluid. A recently introduced viscosity correlation was used to model the effective viscosity of the nanofluid.

The governing equations, continuity equation, momentum equation and energy equation were solved along with the boundary conditions by finite volume method.

The aim of the study is to promote the heat transfer rate using the nanofluid as a working fluid, enhancing the heat transfer can lead to a minimal cost. Various volume fractions were tested in the present study (0% to 6%) at a range of Reynolds number.

It was found that the heat transfer improved considerably with the increase in the volume fraction of the nanoparticle. The heat performance was also promoted with the increase in Re number.

1 Introduction

Heat exchangers are encountered in many industrial applications. Nanofluids can be used in many types of heat exchanger such as those used in power production, the chemical industry, the food industry, heat recovery, refrigeration, and air conditioning [1].

Pantzali et al. [2] presented a numerical and experimental study on the influence of CuO-water nanofluids on plate heat exchanger performance. Their observations revealed that the thermal conductivity augmentation led to significant drop in specific heat and an increase in viscosity. Furthermore, the heat transfer rate was promoted when the flow rate was minimised. Chun et al [3] conducted an experimental investigation on the heat transfer coefficient of three different types of alumina-water nanofluids in a double pipe heat exchanger under laminar flow. The experimental results revealed that an increase in the volume fraction increased the average heat transfer coefficient of the heat exchanger. The researchers reported several key factors that affected this heat transfer enhancement, including nanoparticle size and shape, the properties of nanoparticles, and the volume fraction.
Huminic and Huminic [4] numerically studied the heat transfer performance of double-tube helical heat exchangers using Cu-water and TiO2-water nanofluids under laminar flow. They observed that for CuO-water at volume fraction 2% and same mass flow rate in an inner and outer tube, the heat transfer rate increased by 14% compared to water, as the base fluid. The simulations showed that the increase in the mass flow rate promoted the convective heat transfer coefficients of the nanofluids and water.

An experimental study on the convective heat transfer coefficient of Ag-water nanofluids in a double pipe heat exchanger was presented by Asirvatham et al. [5]. Their results indicated that, under a constant Reynolds number, the nanoparticles significantly enhanced heat transfer. At volume fraction 0.9%, the augmentation in the heat transfer coefficients was as much as 69.3%. The researchers also developed a correlation to predict the Nu number, and then compared experimental results with the results calculated by the correlation; an error of 10% was noted. Mousavi et al [6] studied the effects of a magnetic field on the heat transfer in a double pipe heat exchanger using nanofluids. Their results showed that the inner sinusoidal pipe considerably enhanced the heat transfer rate in the heat exchangers. Their findings showed that the Nusselt number increased by up to 25%. According to their predictions, heat transfer is enhanced by an increase in magnetic field intensity.

Goodarzi et al [7] conducted a numerical and experimental investigation on the heat transfer characteristics of NDG-water nanofluids in a double pipe heat exchanger. They observed an improvement in thermal conductivity of up to 37%. Viscosity decreased with an increase in the temperature by between 51.2 and 51.5%. On the other hand, an increase in Reynolds number and volume fraction could increase the friction factor and thereby increase pressure drop. The main finding was that the key parameters in terms of heat exchanger efficiency enhancement are thermal conductivity and fluid density. Sarafraz et al [8] presented an experimental study on the thermal performance of multi-walled carbon nanotube nanofluids in a double pipe heat exchanger. They observed that that CNT nanofluid promoted convective cooling. The increase in the thermal conductivity due to fluid was found to be 56%, and heat transfer was significantly promoted by this increase in thermal conductivity; however, a marginal pressure drop was also noticed.

Sarafraz and Hormazi [9] investigated the forced convection problem in a double pipe heat exchanger using a novel biological nanofluid. Their results showed that the biological nanofluid changed the flow type from laminar to transient and from transient to turbulent flow. Adding the nanoparticles improved the heat transfer coefficient by up to 67% with almost no penalty in terms of pressure drop. No agreement was found between the experimental results and the available correlations. The authors proposed a new formula in order to predict the heat transfer coefficient, and an average deviation of 17% was found between the experimental data and the proposed model.

Pantzali et al. [10] conducted an experimental study on 16 stainless steel corrugated plate heat exchangers using CuO-water nanofluids, they used water as the hot fluid. Their experimental data showed that the thermophysical properties and flow type inside the heat exchanger played vital roles in the efficiency of the nanofluid as a coolant. They also suggested an empirical correlation to predict the Nu number.

The heat transfer and the pressure drop performance was studied experimentally by Tiwari et al [11]. They measured the thermophysical properties of CeO2-water nanofluid in plate heat exchanger. They reported an enhancement of 39% in heat transfer, and the pressure drop is almost the same as in the case of water. A study on the convective heat transfer in plate heat exchanger using Al2O3-water nanofluid was presented by Pandey et al [12]. They tested the heat transfer and friction factor for volume fractions 2%-4%, the reported an increase in overall heat transfer coefficient up to 110% at Peclet number 7700, where the friction factor decreases with the increase in Peclet number.

Mare´ et al. [13] compared the thermal performance of two nanofluids in a plate heat exchanger in an experimental investigation. Their result showed that alumina and carbon nanotubes demonstrated a better thermal-hydraulic performance and heat transfer enhancement and pumping power loss compared to pure water.

Shakiba et al [14] investigated the effects of a magnetic field on thermal behaviour in a double pipe heat exchanger using Fe3O4-water nanofluids. The results showed that the magnetic field increased the Nusselt number, the friction factor, and the pressure drop. Khoddamrezaee et al [15] presented a numerical simulation on the characteristics of the flow in Al2O3-EG

nanofluid in a shell and tube heat exchanger. Their predictions showed an improvement in the ratio of heat transfer coefficient by up to 3.25%.
Farajollahi et al [16] measured the heat transfer in a shell and tube heat exchanger using Al2O3-water and TiO2-water nanofluids for a wide range of Peclet numbers. They observed an increase in the Nu number when the nanofluids were used compared to water.
Lotfi et al [17] investigated experimentally the heat transfer improvement in a shell and tube heat exchanger utilising multi-walled carbon nanotube water nanofluid for two power heating sections. The maximum enhancement in overall heat transfer coefficient was 6.25% when power heating 630 W was used. Leong et al [18] presented a numerical simulation to study the heat performance in a shell and tube heat exchanger where the cooling fluid is copper water-EG nanofluid. They reported an augmentation in the heat transfer by up to 7.8% when 1% volume fraction of copper-EG nanofluid was used at 26.3 kg/s mass flow rate and 15.97% enhancement in the heat performance when the flow rate increased to 42 kg/s.
Vasu et al [19] presented a parametric study on compact heat exchanger using Al2O3-water nanofluid. They reported that with increase of the coolant inlet temperature, the cooling capacity is increased. Leong et al [20] presented a study on the heat transfer enhancement of a car radiator heat exchanger working with copper-EG. They observed that, about 3.8% of heat transfer enhancement were achieved when 2% copper particles were added in a base fluid at Reynolds number of 6000 and 5000 for air and coolant.
Peyghambarzadeh et al [21] analysed the forced convection in their experimental work when Al2O3-water nanofluids was used in a vehicle radiator under turbulent flow. They observed that with the increase in the fluid circulating rate, the heat transfer performance could be improved. However, they reported an enhancement by up to 45% in heat transfer performance compared to water. Demir et al [22] investigated the forced convection heat performance of Tio2-water nanofluid in a double pipe counter flow heat exchanger. They showed that the surface heat transfer coefficient increased when 4% volume fraction was used compared to water, however, they reported an increase in pressure drop by up to 16%.
Zamzamian et al [23] studied the turbulent flow forced convective heat transfer performance in a double pipe and plate heat exchangers with Al2O3-ethylene glycol and CuO-ethylene glycol. Their measurement data showed that the maximum heat transfer enhancement was 49% and the lowest enhancement was 3%. However, they reported a considerable discrepancy at higher temperatures and volume fractions. Haghshenas et al [24] investigated the flow in tube and plate heat exchanger with ZnO-water nanofluid. They found that in the plate heat exchanger, the heat transfer coefficient of ZnO-water nanofluid at mass flow rate 10 g/s was nearly 20% higher than water and under the same conditions in the concentric heat exchanger is 14% higher than water.
Kwon et al. [25] conducted an experimental work on the heat transfer characteristics and pressure drop of the ZnO and Al_2O_3 nanofluids in a plate heat exchanger. They found that he overall heat transfer coefficient at the same Re number increased by up to 30% at 6 % volume fraction for Al2O3-water nanofluid in comparison to water. Duangthongsuk and Wongwises [26] investigated experimentally the heat performance and the pressure drop of TiO2 water nanofluid in a double pipe counter flow heat exchanger. The data showed that the heat transfer coefficient is higher than that water and promoted with the rise in Reynolds number and it was found 26% higher than that the base fluid. The results also showed that the pressure drop was slightly higher than the base fluid and increased with the increase in the volume fraction.
Yang et al [27] investigated the convective heat transfer coefficient of graphite nanofluid flowing in a horizontal tube heat exchanger. Their finding showed that the heat transfer coefficient improved with the increase in Re number as well the volume fraction. They presented a new correlation to predict the heat transfer coefficient under laminar flow. Kumar et al [28] conducted an experimental investigation to study the heat transfer and friction for Fe3O4-water nanofluid in a double pipe heat exchanger. In their results, the Nusselt number improved by up to 14.7% at

volume fraction 0.06% at a Re number of 30,000 in comparison to water. However, they reported a penalty of less 10% in the pumping power. They proposed new correlations to predict Nusselt number and friction factor.

Shirvan et al [29] studied the effectiveness of the heat transfer in a double pipe heat exchanger using Al2O3-water nanofluid. Their results showed an enhancement in Nu number by 57.7% at volume fraction 3% for Re number range 50 to 150. Doruk et al [30] investigated experimentally the heat performance and pressure drop of nanoencapsulated water nanofluid in a double pipe heat exchanger. In their results, they observed no improvement in the heat transfer coefficient for the nanofluids at volume fractions 0.42% and 0.84%. However, they reported an enhancement of 10% the nanofluids at volume concentration 1.68%.

The aim of the present work is to investigate the enhancement of heat transfer in a shell and tube heat exchanger with inclined baffles using a titanium oxide-water nanofluid as a cooling fluid.

2 Problem Description

A schematic diagram of the geometry is shown in Fig.1. The model considered is a shell and tube heat exchanger, the heat exchanger is supplied with five horizontal pipes with a diameter 1 cm, the diameter of the shell is 10 cm, the diameters of the water inlet and outlet are 3 cm, the length of the heat exchanger is 30 cm. Water enters the shell with constant velocity temperature 360 K and leaves at zero pressure gauge , while titanium-oxide water nanofluid enters 5 tubes with constant axial velocity and temperature 300 K the upper and bottom walls are thermally insulated, the left wall is heated at temperature T_H and the right wall is maintained at lower temperature T_C. The enclosure is filled with water based nanofluid, the nanoparticle investigated is TiO_2 with a spherical diameter of 25 nm. The Thermal properties are listed in Table 1

Table 1 Thermophysical properties of the nanoparticle

Nanoparticle	Density Kg/m^3	Thermal conductivity w.m^{-1}.k^{-1}	Specific heat J.kg^{-1}.k^{-1}
TiO2	4250	8.93	686.2

The governing equations were solved in the present study are continuity, momentum equation and energy equation and can be written as

Continuity equation

$$\nabla(\rho\vec{v}) = 0 \quad (1)$$

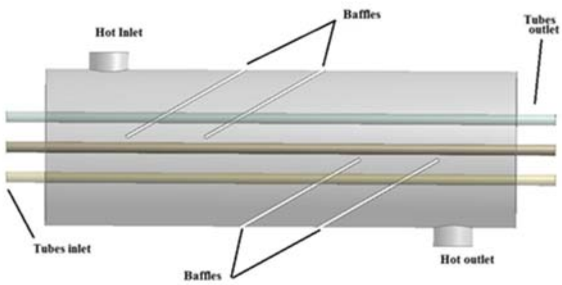

Figure 1 Shell and tube 3D geometry

Momentum equation

$$\nabla(\rho\vec{v})\vec{v} = -\nabla p + \nabla(\tau) + \rho g + F \quad (2)$$

Energy equation

$$\nabla(\vec{v}(\rho E + P)) = \nabla\left(K_{eff}\nabla T - \sum h_j J_j + (\tau\vec{v})\right) \quad (3)$$

The density of Nanofluid is expressed as:

$$\rho_{nf} = (1-\phi)\rho_f + \emptyset\rho_s \quad (4)$$

The specific heat of the nanofluid is expressed as:

$$C_{pnf} = \frac{(1-\phi)(\rho C_p)_f + \phi(\rho C_p)_s}{\rho_{nf}} \quad (5)$$

The thermal conductivity can be calculated as:

$$K_{eff} = K_f\left(\frac{K_s + 2K_f - 2\emptyset(K_f - k_s)}{K_s + 2K_f + \emptyset(K_f - k_s)}\right) \quad (6)$$

The effective viscosity is written as [31]:

$$\mu_{eff} = \mu_f(1 + 5\emptyset + 80\emptyset^2 + 160\emptyset^3) \quad (7)$$

The heat transfer coefficient can be written as:

$$h = \frac{Q}{(T_H - T_C)} \quad (8)$$

Nu number is calculated as:

$$Nu = \frac{hL}{k} \quad (9)$$

Re number is expressed as:

$$Re = \frac{UD\rho}{\mu} \quad (10)$$

2.1 Numerical Procedure

The CFD code used in this investigation is ANSYS 15. The nanofluid is modelled as single phase model. Equations (4), (5), (6) and (7) have been used to model the density, specific heat, thermal conductivity and viscosity respectively. The turbulent model K-Ɛ was employed. For pressure velocity coupling, Courant number=100, under relaxation factor was chosen 1 for density, body force and energy. Explicit relaxation factor 0.5 for momentum and pressure, body force weighted for pressure spatial discretization, the time step=0.02 s, number of time steps=1000, the transient formulation is first order implicit.

2.1. Grid dependency test

A grid independence test was carried out in order to ensure that the solution obtained is mesh independent. The simulations were first done for various meshes with different number of cells and the average Nu was calculated for each mesh. The five meshes tested with Nu results are shown in Table 2. As it can be seen that Nu number for mesh 4 with 375640 cells found remain unchanged, and hence mesh 4 was selected for the present numerical simulations

Mesh	Number of cells	Nu
Mesh 1	225500	44.2
Mesh 2	305460	58.41
Mesh 3	345840	61.23
Mesh 4	375640	62.05
Mesh 5	412226	62.11

3 Results and discussion

In this section, the results are introduced and discussed; the change of the temperature in the centre line of a tube is shown in Figure 2.

The Figure illustrates the change of temperature for volume fraction 2%, 4% and water. It can be seen that the temperature is lower at the tube inlet (300 K) and increases rapidly due to the exchange of heat as a result of higher temperature from the hot water inlet (360 K).

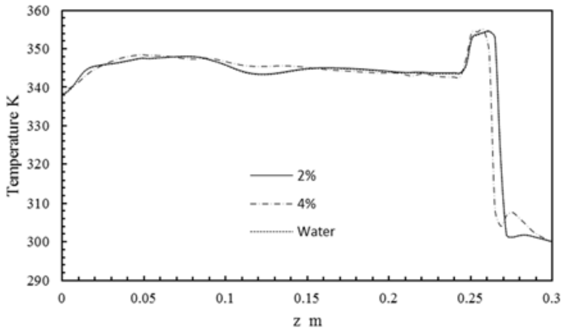

Figure 2 Temperature variation with volume fraction at tube wall

The change of viscosity at the centre of the tube is presented in Figure 3. It is evident that the viscosity of the nanofluid is higher at inlet where the temperature is lower. It should be mentioned that the viscosity predicted in the present work is temperature dependent; the same applies to thermal conductivity, specific heat and density. It can also be seen clearly that the viscosity increases with the increase of the nanoparticle volume fraction.

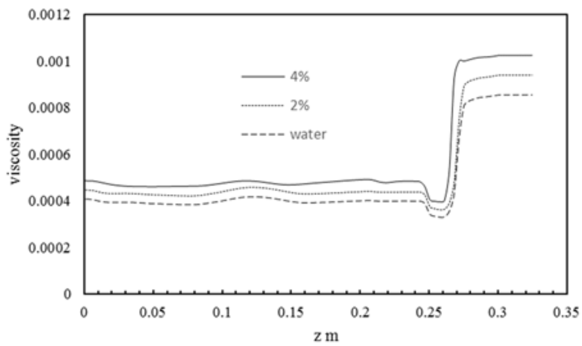

Figure 3 Viscosity change with volume fraction at the tube

The pressure coefficient is studied for the range of volume fraction along the tube tested. The results are shown in Figure 4. The results depicted for a tube in the heat exchanger at a horizontal axis, which shows the rise of the pressure coefficient. It was found that the pressure increases substantially at the inlet and becomes steady along the tube.

It was also noticed that the pressure coefficient increases with the increase in the nanoparticle volume fraction.

The streamlines of the velocity of the flow entering the shell is illustrated in Figure 4. It can be seen that the hot water enters the shell with a high velocity and due to the fins and the tubes, the velocity decreases massively as a results of the vortex in the flow domain.

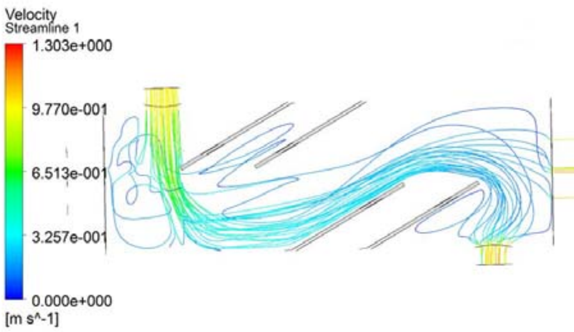

Figure 4 velocity streamlines in the shell

The contours of the temperature for three tubes are shown in Figure 5. It is obvious that the temperature enters the tubes with a low temperature 300 K and due to the hot water in the shell, the temperature jumped considerably as captured in the Figure.

It should also be mentioned that the highest temperatures can be found in the region where the hot water enters the shell which is expected to exchange the temperature and this represents the highest temepraturе at the tubes wall.

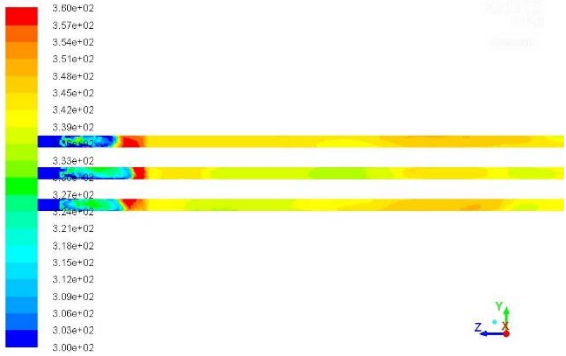

Figure 5 Temperature contours in the tubes

The thermal conductivity contours for the three tubes are also studied and the contours are introduced in Figure 5. It should be mentioned that the contours are for volume fraction 2%.

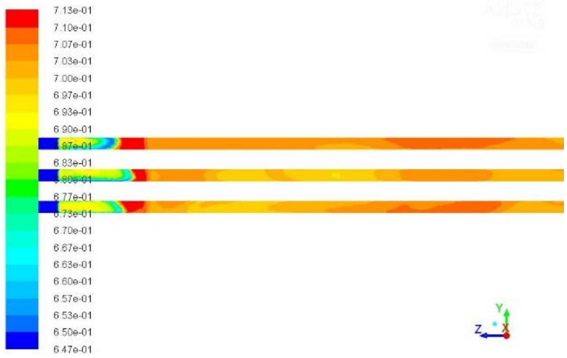

Figure 6 Thermal conductivity contours for tubes

The viscosity contours are also shown in Figure 7. It was found that the viscosity of the nanofluid for 2 % volume fraction is higher at the nanofluid inlet to the tubes, the viscosity drops gradually due to the rise in the temperature in the tubes. The viscosity decrease in noticed in the hot water inlet location, whereas the viscosity starts increasing again due to the decrease of the temperature at that point onwards.

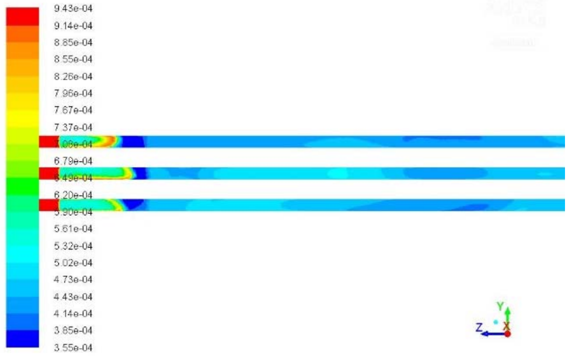

Figure 7 viscosity contours for tubes

The streamlines of the velocity and the temperature at the shell and tubes is depicted in Figure 8. It is shown that the velocity is lower at regions where the temperature is higher in the tubes. This is in line with the temperature contours and velocity depicted in Fgure 4.

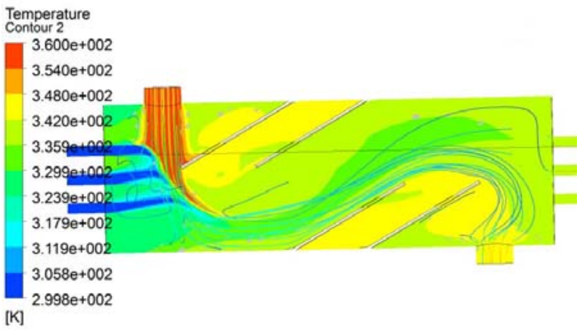

Figure 8 streamlines and temperature contours.

To give a better understanding of the flow inside the shell, Figure 9 is introduced and it illustrates the distribution of the temperature of the hot water at the inlet of the shell. It is obvious that the hot fluid enters with a high temperature and the flow circulate in the shell, as a result to that the temperature drops considerably

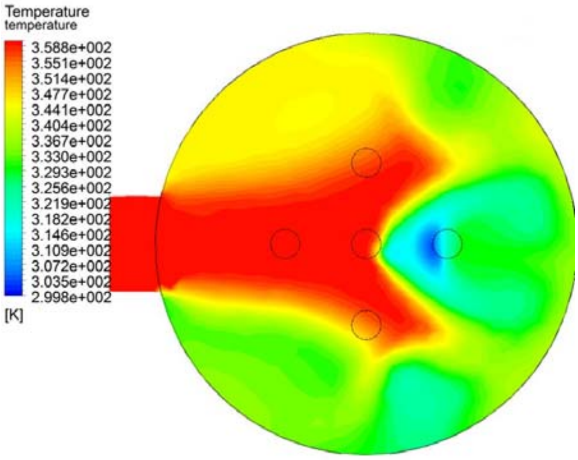

Figure 9 Temperature distribution of the hot fluid inlet

4 Conclusion

- The fluid flow characteristics and heat transfer flow is investigated for TiO2-water nanofluid in a shell and tube heat exchanger.
- The results showed that the thermal conductivity increased with the increase in the volume fraction of the nanoparticle.
- The temperature of the hot fluid decreases with the advance of the flow in the shell.
- The pressure coefficient increases rapidly at the inlet and remains steady in the tubes.

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